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# Development and Evaluation of Titanium Spacesuit Bearings

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The Z-2 Prototype Planetary Extravehicular Space Suit Assembly is a continuation of NASA's Z-series of spacesuits, designed with the intent of meeting a wide variety of exploration mission objectives, including human exploration of the Martian surface. Incorporating titanium bearings into the Z-series space suit architecture allows us to reduce mass by an estimated 23 lbs per suit system compared to the previously used stainless steel bearing race designs, without compromising suit functionality. There are two obstacles to overcome when using titanium for a bearing race- 1) titanium is flammable when exposed to the oxygen wetted environment inside the space suit and 2) titanium's poor wear properties are often challenging to overcome in tribology applications. In order to evaluate the ignitability of a titanium space suit bearing, a series of tests were conducted at White Sands Test Facility (WSTF) that introduced the bearings to an extreme test profile, with multiple failures imbedded into the test bearings. The testing showed no signs of ignition in the most extreme test cases; however, substantial wear of the bearing races was observed. In order to design a bearing that can last an entire exploration mission (~3 years), design parameters for maximum contact stress need to be identified. To identify these design parameters, bearing test rigs were developed that allow for the quick evaluation of various bearing ball loads, ball diameters, lubricants, and surface treatments. This test data will allow designers to minimize the titanium bearing mass for a specific material and lubricant combination and design around a cycle life requirement for an exploration mission. This paper reviews the current research and testing that has been performed on titanium bearing races to evaluate the use of such materials in an enriched oxygen environment and to optimize the bearing assembly mass and tribological properties to accommodate for the high bearing cycle life for an exploration mission.

## Nomenclature

$\angle_O$	=	angular movement of outer race, degrees
$\angle_B$	=	angular movement of bearing ball axis, degrees
$I_{CIRC}$	=	inner race circumference
$O_{CIRC}$	=	outer race circumference

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## I. Introduction

Since the increased availability of titanium alloys over the past century, many product designs have included these alloys due to their relatively light weight (37% weight reduction compared to a cast iron component of same dimensions [1]), high corrosion resistance, and biocompatibility [2–4]. Applications for these alloys have included biomedical implants, such as orthopaedic devices and dental implants [2, 5, 6], aircraft and aerospace components such as bolted interfaces, pinned joints, and couplings in structural and engine assemblies [7], automotive brake pads [1] and valves in chemical process industry [3]. One application that NASA has been investigating is the use of titanium bearing systems over stainless steel in advanced planetary spacesuits. System mass is extremely important for future spacesuit systems and the use of Ti-6Al-4V for the bearing races, as opposed to the traditional stainless steel, can save an estimated 23 lbs on the suit system.

Although titanium and its alloys (most common is Ti-6Al-4V [2, 3]) would be an ideal candidate for many applications based on its reduced density and corrosion resistance, there are two main problems with using titanium for spacesuit bearings, (1) titanium is flammable in 100% oxygen environment [21] and (2) many studies have also shown that these materials have poor wear resistance [1–4]. Previous studies have shown that titanium is flammable in a 100% oxygen environment at pressures as low as 1 psia [21], but there are also some studies showing limited ability to create ignition of the titanium races in bearing systems [19]. As a result, NASA executed a test series to evaluate if titanium bearings from multiple locations with multiple failures simulated would pose an ignition risk [20].

Titanium alloys poor tribological properties can be accredited to two main factors- (1) titanium's low resistance to plastic shearing and low work hardening [4] and (2) the low protection of the underlying metal by the surface oxide, which forms due to high flash temperatures induced by friction during dry sliding [4]. During wear, the oxide layer that naturally forms (usually  $\text{TiO}_2$ ) is easily worn away by spalling or micro-fragmentation. This native oxide is slow to reform, which results in an unprotected surface more vulnerable to more severe wear. Not only does the removal of the oxide layer allow for more wear, the dissolved oxygen coming from the atmosphere tends to embrittle the matrix, reducing the mechanical resistance of the material [8]. The issue with the spacesuit, is that the race is can be exposed to low pressure and likely vacuum conditions during operation, limiting the opportunity for re-oxidation of the titanium surface to improve wear resistance. Due to this reduced ability to oxidize, lubricants and surface coatings will be needed to improve the wear resistance of the titanium races.

The main causes of wear and failure in bearing races are excessive loads, overheating, true brinelling, fatigue, contamination, misalignment, corrosion, and lubricant failure [9]. For the case of vacuum sealed bearing races, lubricant failure plays a large role in the wear of the race system. Since the lubricant cannot be constantly replenished to remove wear debris and keep the bearings in an elastohydrodynamic state, the lubricant has the potential to dry and create more sliding wear in the bearing system.

The wear mechanism that is the driving force for the poor sliding wear in titanium is galling [10]. Wear mechanisms describe how the material is being removed from the surface. Multiple wear mechanisms will occur simultaneously, but one mode will tend to dominate during different stages of the wear process [11, 12]. Galling is a severe form of adhesive wear between coupling solids, where material from the worn area flow up from one or both surfaces and adherence occurs [13, 14]. With high loads and poor lubrication, when the wear system galls, seizure often takes place. Galling usually occurs in face centered cubic (f.c.c.) metals due to increased cross-slip than in other crystal structures [14]. Hexagonal closed packed (h.c.p.) metals with a high  $c/a$  ratio have a low dislocation cross slip rate and are less prone to galling, but titanium's ratio is lower, which allows for an increased chance to gall. It should be noted that not all researchers agree that galling is an issue for titanium alloys. When Budinski *et al.* studied a galling-resistant substitute for a silicon nickel alloy, they found that materials mated with Ti-6Al-4V and CP (commercially pure) Ti showed the best resistance to galling (or highest threshold galling stress) [15]. They also showed that Ti-6Al-4V is more galling resistance than CP Ti.

Literature has shown studies of researchers attempting to alter the material or tribosystem to prevent galling. The amount of galling can potentially be prevented by the following six techniques: (1) allowing for sufficient space between the two components in the wear system (tightly packed components are prone to galling) [16], (2) using adequate lubrication to separate the components [17], (3) coating the components [17], (4) increasing the hardness of the tribosystem component [14], (5) controlling the surface roughness (highly and rough finishes increase tendency to gall) [16], and (6) altering the composition of the opposing surfaces [3].

## II. Ignitability Testing of Space Suit Bearing Systems

Ti6Al4V is flammable in 100% oxygen at pressures as low as 1 psia [21]. Previous testing was conducted to evaluate the use of Titanium bearing races, and proved that by sliding a rotator ball set over a stator race while in spacesuit pressures, sparks could be created [18]. However, additional testing was executed on wrist bearings to

evaluate whether or not a realistic bearing system was ignitable [19]. To build on this previous testing and to test the ignitability of other suit bearing assemblies, bearing designs from through out the suit assembly were chosen based on the extremes for cycle speed, cycle distance, expected cycle life required, and the load on the bearing system [20]. Choosing bearing systems that maximized each of these factors allows the test results to encompass all other bearings on the suit.

To design a conservative test, two simulated failures were incorporated into the test: (1) a primary seal failure and (2) an obstruction in the race in the form of a mis-matched ball port. Nominally, the bearing race is sealed from the suit environment; however, there is a secondary seal outside of the race. As a result, if the primary seal fails and the failure is not identified, the race will see the suit oxygen concentration and pressure. The test was conducted at higher oxygen pressure (12.4 psia) to add conservatism and simplify the test setup. Additionally, an obstruction in the form of a mis-matched ball port was added to create a large bump in the race. The bearings were tested for 96 hours while torque, load, temperature and cycle rate were recorded. Summary of the test conditions is shown in Table 1.



**Figure 1: Hip Bearing Ignitability Cycle Test Results**

Testing showed no signs of ignition on the bearings, but displayed significant wear on the hip bearing (Figure 1) and more moderate wear was seen on the waist bearing. Test results highlighted the wear problems with titanium bearings and initiated work to investigate design requirements for the material.

**Table 1: WSTF Ignitability Test Matrix**

Bearing Location	Cycle Speed	Total Load, lbf	Cycle Rate
Hip	45° @ 78 %s	1983	20/min @ 1hr, 40/min @ 45min, 52/min @ 30min
Shoulder	125° @ 135 %s	676	20/hr
Waist	30° @ 52 %s	1111	20/min @ 1hr, 40/min @ 45min, 52/min @ 30min

### III. Titanium Bearing Contact Stress Analysis

#### A. Bearing Locations Studied

This effort focuses on the same three suit bearing locations used at the WSTF testing for the development of contact pressure design limits. The bearings chosen are part of NASA's Z-1 Advanced Space Suit and are located at the waist, hip, and scye (shoulder). These specific bearings are real-world examples of titanium bearings used in a space suit, and have well-defined geometries, loads, and performance data. All three bearings use Ti-6Al-4V races, and stainless steel 440C bearing balls. The stainless steel balls have smaller, plastic balls sitting between them in the groove; these separator balls do not carry the bearing load but serve to keep rolling friction down and preserve steel ball spacing.

**Table 2: Bearing location information**

Bearing Location	Total Load, lbf	Number of Steel Balls	Ball Dia, in	Cycle Arc, deg	Number of Cycles
Waist	1983	100	0.250	60	200,000
Scye	676	78	0.187	250	200,000
Hip	1111	142	0.131	90	200,000

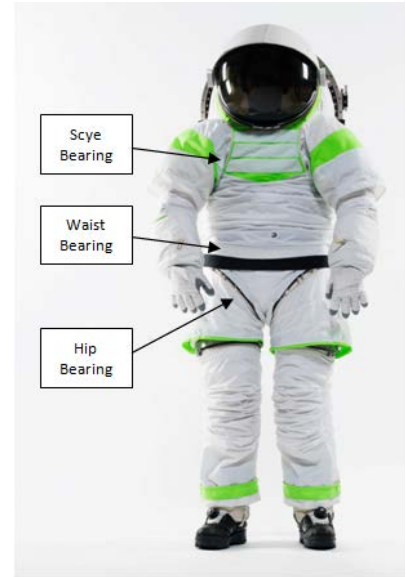
## B. Contact Stress Calculation

Contact stress is the study of how contacting bodies exert pressure on one another. As two bodies come into contact and mutually deform, the shape and size of the shared contact zone becomes difficult to calculate. Simplified techniques exist for the derivation of the contact zone, and the peak pressures that result in each body. In the interest of determining the stresses shared between bearing ball and race during rotation, the “ball-in-groove” adaptation of Hertz’s technique is well suited. This geometry approximates the round groove inscribed around the bearing race as a cylinder of infinite length.

The inputs for a ball-in-groove stress calculation are the forces between the ball and the race, the orthogonal radii of the two bodies, and the material properties of the ball and bearing race. The two radii of the bearing ball are both equal due to the symmetry of a sphere. The groove however, has one defined radii (cross sectional shape) and an infinite radii in the other direction.

While the contact stress calculation only studies a single bearing ball, actual bearings consist of many balls. The force used in the contact stress calculation is a fraction of the total bearing load, and depends on how this total load is shared amongst all the balls. The test samples used for this study are thrust bearings, and their configuration ensures the total bearing load is shared equally amongst all bearing balls. This allows for a simple relationship between bearing load, ball load, and contact stress for a test sample, and ensures accurate relationships between the test variable and the test results.

It is important to note that for future implementation of the contact stress limits developed in this study, a more complex relationship exists between spacesuit bearing total load and individual ball load. The total bearing load is not uniformly distributed among all the bearing balls in a suit bearing. Spacesuit bearings are attached to axial restraint lines, which carry a significant portion of the suit loading to help relieve tension in the suit fabric. These restraint lines attach to the bearing races at localized points, creating force concentrations and uneven race loading. Race stiffness also varies at certain points due to mounting holes, access plugs, and locking mechanisms at various locations. Studies must be performed to determine the highest loaded ball in each bearing application and apply the stress limit to that location. If we assume a perfectly even load distribution in the actual bearings, the contact stress values are as shown below.



**Figure 2: Z-1 spacesuit bearings**

**Table 3: Contact stress in Z1 bearings assuming balls evenly loaded**

Bearing	Load/Ball, lbf	Stainless 440c Ball		Titanium 6Al-4V Race		Contact Stress, ksi	Contact Stress, % of Yield
		Poisson's Ratio	Modulus, ksi	Poisson's Ratio	Modulus, ksi		
Waist	28.0	0.3	29,000	0.342	16,500	164	113
Scye	12.3	0.3	29,000	0.342	16,500	151	104
Hip	11.1	0.3	29,000	0.342	16,500	183	126

Another consideration when applying test results to spacesuit bearings is that in the test bearings, no load multiplication occurs when force is transferred from one race, through a ball, to the other race. This is the nature of a thrust bearing configuration. Spacesuit bearings use a radial bearing configuration, and when loaded in thrust, the load between races is transmitted through the ball at an angle. This bearing contact angle creates a force multiplication from the inclined-plane-type geometry. While nominal contact angle and resulting load increase can be calculated, non-uniform loading and deflections in the bearing races can alter this contact angle for each ball.

## IV. Bearing Cycle Test Plan

### A. Test Bearing Assemblies

Testing for this effort will require cycling a bearing at a set load and documenting it's condition after a fixed test duration. The goal is to have several worn samples to observe trends between load, stress, and degree of failure. Due to the size and complexity of the subject spacesuit bearings, manufacturing many of each type for testing is not economical. Three low-cost bearings were designed for use as test subjects, all of which utilized key spacesuit bearing geometry in a smaller and simpler package.

A thrust bearing arrangement was chosen over the suit-style radial ball bearing. Since the suit bearing is loaded in thrust, the balls contact the race at an angle close to 45°. The thrust bearing test samples will have a vertical line of contact between the balls and races. A common outside diameter (OD), overall height, and mounting provision was used for all three sample designs. All samples are made from Ti-6Al-4V. The same ball sizes and corresponding groove radii were carried over from the spacesuit bearings to the test samples. To package the ball complement into the 3" OD test sample, pitch diameters were assigned that offer clearances equivalent to the parent bearings. Each bearing is comprised of two identical races.



Figure 3: Test bearing races

### B. Test Duration

Since we are using a smaller bearing to simulate the larger spacesuit bearing, it was necessary to determine how much to cycle the small bearing so it experiences the same fatigue as the real bearing over its lifetime. Bearing fatigue occurs each time a ball passes over the same spot on the bearing race; we call these occurrences "stress events". If the same number of stress events occurs in both the sample bearing and the real bearing, they are equally fatigued. As an example, if 100 bearing balls pass over the same spot in a race during one cycle, this would be considered 100 stress events. If the desired life of this same bearing is 200,000 cycles, then the desired life can be expressed as 20,000,000 stress events (200,000 cycles x 100 stress events). In a smaller bearing with fewer balls, to achieve 20,000,000 stress events it will require more than 200,000 cycles.

To determine the number of times the same spot in a bearing race is repeatedly loaded and unloaded during a cycle, the relative motion of the outer race, balls, and inner race must be understood. In a thrust bearing configuration, as with the test plates, this is a simple relationship because both races contact the balls along the same diameter, so the rolling path is the same. The ball travels the same amount relative to either race for a given movement of the bearing. In an angular contact bearing like those used in spacesuits, the relative movement of the races and balls is more complex. Because both races contact the ball along a different diameter, the rolling paths are different, and the ball travels further on one race than on the other during bearing rotation. Calculations revealed the angular movement of each ball can be expressed as a function of the angular movement of the outer race, the inner race contact circumference, and outer race contact circumference:

$$\angle_O = \angle_B \left[ 1 + \frac{I_{CIRC}}{O_{CIRC}} \right] \quad (1)$$

Once the angular movement of the balls is known for a given race movement, the number of balls laying in that arc reveals how many repeat contacts occur on the inner race and the outer race (arc size is  $\angle_O - \angle_B$ ). For example, in a 250° cycle of the scye bearing, the balls move 126.819° relative to the inner race, and 123.181° relative to the outer race. If it were a thrust bearing arrangement with the same cycle size, the relative motion of the balls would be 125° to either race.

For this study, the focus is on the scye bearing because this location undergoes more cycles than the other two. This is the most demanding application and the design limits discovered here can be applied to the other locations with safety. We will arrive at contact pressure limits for three different ball sizes when installed in a scye-sized bearing. Each ball size will have a different number of stress events after the same number of cycles. This will shed light on the impact of changing ball size to reduce contact pressure, a viable option if analysis reveals contact stresses are too high. Table 4 summarizes the test durations for each sample bearing.



**Table 4: Test Plate Rotations**

Ball Size, in	Plate Pitch $\emptyset$ , in	Number of Balls	Desired Stress Events	Plate Rotations	Speed, rpm	Hours Required
0.131	2.4593	30	7,855,524	523,702	400	21.8
0.187	2.4817	21	5,495,476	523,379	400	21.8
0.250	2.4864	16	4,176,341	522,043	400	21.8

Several factors were considered when choosing the speed for test plate rotation. Rotation speed must allow for short test times without inducing any dynamic effects that could influence bearing life. Very high speeds could potentially raise temperatures, increase radial loading, diminish lubrication, and induce vibration. Linear speed of ball position was evaluated for each suit bearing to understand the operating condition. The plate rotation speed was set at 400 revolutions per minute. Future testing will evaluate the effects of slower cycles and changing direction of motion during cycling.

### C. Design of Test Stand

To conduct titanium bearing testing a machine was designed that could carry out unsupervised tests while recording and monitoring bearing performance data. The test machine is instrumented and computer controlled so it could continuously capture data, with the capability for automatic shut off at point of bearing failure. This test system can be broken into three parts; test machine, control devices, and instrumentation.

#### 1. Test Machine

The test machine is designed to rotate a test bearing along a vertical axis while applying a vertical load. The machine must apply this load in a steady manner with good alignment, so internal loads in the bearing are evenly distributed and predictable. Rotational motion must also be steady and with good alignment, so no stray radial loads are applied to the bearing. The test machine utilizes several key principles:

- Top bearing race is fixed to establish bearing centerline, bottom race aligns to top race axis.
- Bottom bearing race rides on floating thrust bearing to establish horizontal plane. Top plate has ability to pivot and align to bottom race plane.
- An air cylinder pivots a load arm; the arm contacts the top bearing race at the center of its length to apply normal load.
- A needle roller bearing transfers force from arm to bearing; the roller ensures movement of arm does not apply transverse loads to bearing.
- The load arm applies a symmetric load across the thick aluminum machine deck.

**Figure 4: Test machine**

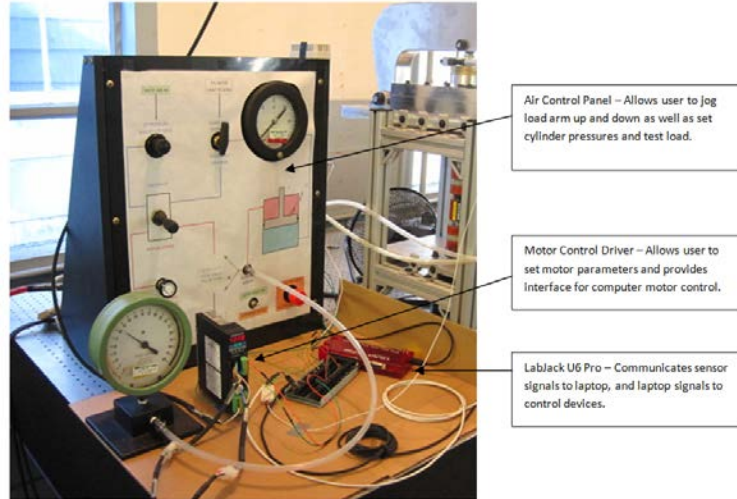
#### 2. Control Devices

Several devices are used to convert human and computer control signals to machine actions. To enable automatic stop of the machine at end of test or at bearing failure, the motor must be stopped and the air cylinder must be relieved of pressure. The stand-alone motor control driver is connected to the computer interface to gain control of motor operation. To control the air cylinder, a pneumatic system was designed and attached to a panel. This system would allow for human control of arm up and down, as well as computer controlled relief of downward pressure.

### 3. Instrumentation

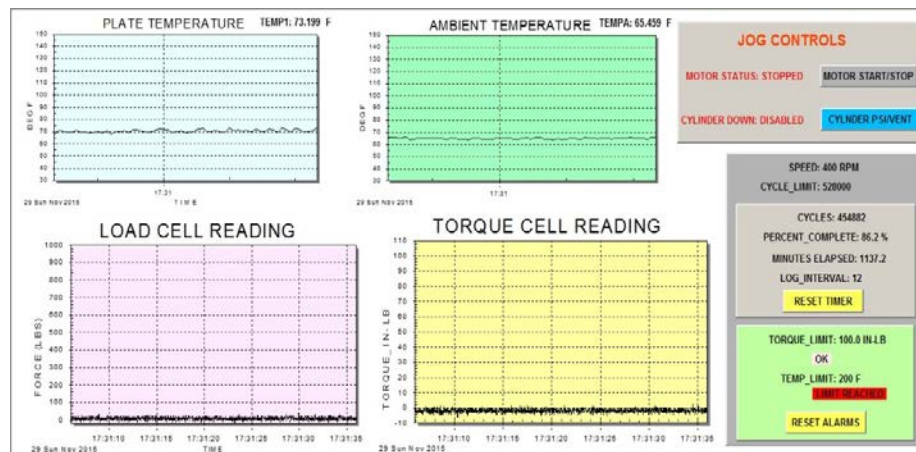
The machine is equipped with four sensors to collect test data; a load cell, torque cell, plate temperature sensor, and ambient temperature sensor. A description of each sensor is provided below.

- Load Cell – Donut Style, 3000 lb capacity, 4 wire, 2mv/v, 1.48" OD, by Futek
- Torque Cell – Shaft to Shaft Rotary Style, 440 in-lb capacity, 5 wire, by Futek
- Temp Sensors – Both RTD Style, 3 wire, plate uses disc type sensor on top race



**Figure 5: Control devices**

A USB data acquisition interface is used to communicate the sensor signals to the laptop, and the laptop signals to the control devices. Our interface is a LabJack U6 pro interface. The signals sent from the LabJack to the laptop are then interpreted, displayed, and logged by DAQFactory express data acquisition software. A screen-shot of the DAQFactory interface created for bearing testing is shown in Figure 6.



**Figure 6: Instrumentation screen.**

Due to the long test times, thought was given to the amount of data that was practical to store for each test. Our strategy was to collect one second of sensor readings every minute for the entire length of the test. We believe that changes in performance would be slow enough that nothing drastic would occur and be missed between minutes. Within that one second of recording, many individual readings can be taken from each sensor. We chose to take 50 readings per second for both the load cell and torque cell, and only take 1 reading per second for both temp sensors. At this rate, each twenty-two hour test would generate 66,000 rows of sensor data, a set small enough to be handled in Microsoft Excel.

### D. Execution of Tests

Testing starts with the baseline samples for each of the three ball sizes. The duration of each test is twenty-two hours, which conveniently allows for the next test to begin at the same time the next day. The two spare hours in between are useful for test setup and data organization. Time is allotted for five tests of each ball size; this provides enough data to determine a pass/fail boundary. After the baseline tests are complete, the modified assemblies are tested in the same fashion.

The test machine runs continuously and automatically stops when either desired life is achieved or performance measurements exceed maximum values. Maximum allowable test temperature is set at 200 °F and maximum test torque at 150 in-lbs. At end of test, samples are photographed on and off of the test machine to document any debris or dust produced during the test. Next the plates are disassembled, photographed, and cleaned. Cleaned components are observed and photographed under magnification to determine if material breakdown occurred. Visual inspection combined with review of the logged test data will result in a pass/fail assignment for the test. If the sample failed, contact pressure is reduced for the next test. If failure did not occur, contact pressure is increased for the next test to try and arrive at the failure point.

## V. Test Results

After completion of testing each bearing was evaluated as either a pass or fail based on the amount of wear seen. Table 5 which lists each test in the order it was performed and indicates the test settings, pass or fail outcome, and contact pressure.

**Table 5: Summary of the bearing results.**

Ball	Result	Plate Load (lbs)	Contact Stress (%Ti Yield)	Contact Stress Value (psi)
440C - Ø.187	Pass	229	100%	145,000
	Fail	772	150%	217,500
	Fail	447	125%	181,250
	Pass	305	110%	159,500
440C - Ø.131	Pass	165	100%	145,000
	Fail	322	125%	181,250
	Fail	251	115%	166,750
440C - Ø.250	Pass	310	100%	145,000
	Pass/Fail	605	125%	181,250
	Fail	761	135%	195,750

The following information is provided as an example of the type of data collected and the distinction between passing and failing samples. Results are shown for the 0.250" ball diameter test bearings. A progression towards bearing failure is seen as load is increased from 100% to 125% to 135% of titanium yield strength.



**Figure 7: Test bearing, 0.25 inch ball, 100% load**

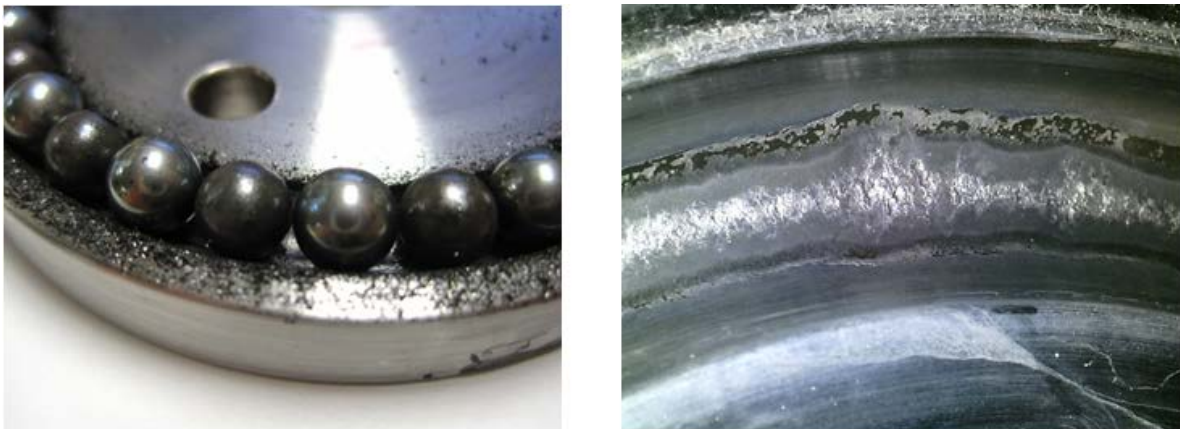
Figure 7 shows condition of races after testing when loaded to 100% Ti yield strength. Note lack of debris in the left picture and uniform finish on the ball groove in the right picture. This bearing was loaded to 310 lbs for 528,000 cycles.





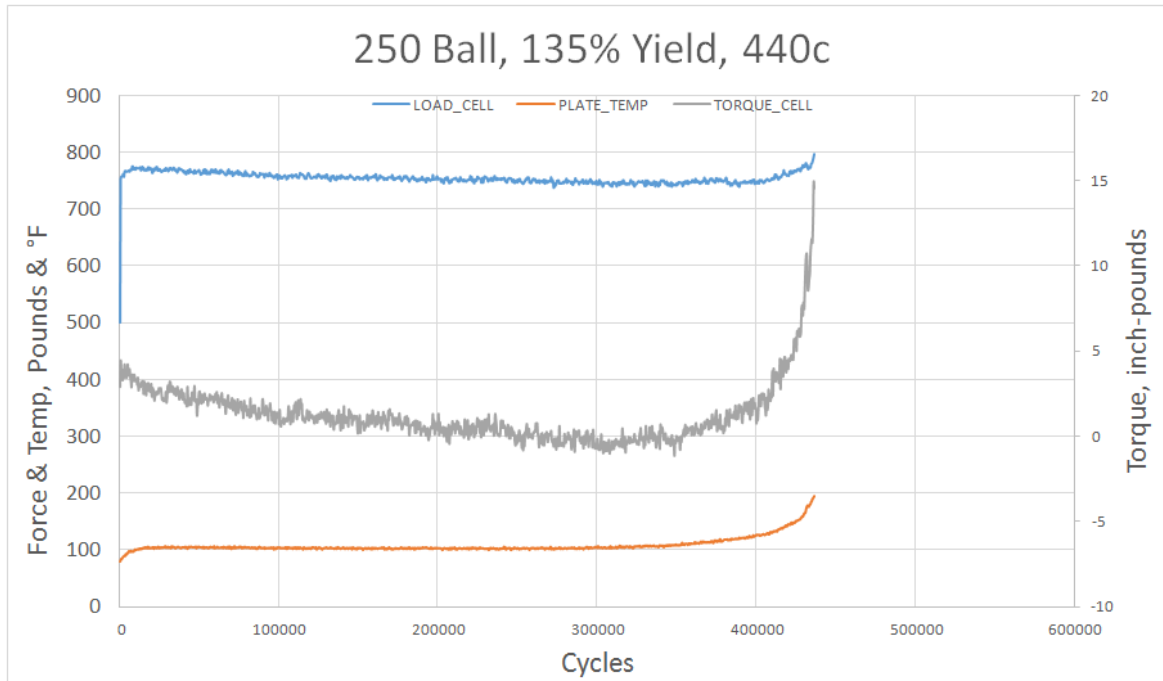
**Figure 8: Test bearing, 0.25 inch ball, 125% load.**

Figure 8 shows condition of races after testing when loaded to 125% Ti yield strength. This bearing was loaded to 605 lbs for 528,000 cycles. There was little, if any, metal debris generated, but the presence of black titanium “dust” is a precursor to failure. Magnification of the races shows a few localized failure points centered directly on the ball contact path. While the presence of these flaws didn’t dramatically alter performance, it is believe this bearing material is failing due to fatigue.



**Figure 9: Test bearing, 0.25 inch ball, 135% load**

Figure 9 shows condition of races after testing when loaded to 135% Ti yield strength. This bearing was loaded to 761 lbs for 430,000 cycles. Presence of dust and metal flakes are an indication of considerate race breakdown. Magnification of the groove shows significant material failure, with damage throughout the contact path. These results show a dramatic change in survival of the bearing from 125% to 135% contact stress loading.



**Figure 10: Test data, 0.25 inch ball, 135% load.**

A graph of the performance data from the 135% test is included shown in Figure 10. Material failure coincides with a sharp rise in both plate temperature (orange line) and running torque (gray line). Performance data from a surviving bearing is steady throughout the duration of the test. Although it looks like torque values actually decrease in the middle of the test, this is a result of the sensor reacting to small ambient temperature changes. This test was stopped early at around 430,000 cycles (81% of total life) due to high plate temperatures.

## VI. Conclusion

After completion of bearing testing the following guidelines were developed:

**Table 6: Contact stress guidelines**

Titanium Race Design Guidelines for Contact Stress and the Cycle Life Required	
Titanium races have excellent resistance to contact stress damage when kept below 100% of titanium's yield strength (145 ksi). This should be used as a design consideration for nominal long term operation of a bearing.	
Titanium races have mild resistance to contact stress damage when kept between 100% of titanium's yield strength (145 ksi) and 115% of titanium's yield strength (165 ksi). This should be used as a maximum design limit for nominal long term operation of a bearing.	
Titanium races have poor resistance to contact stress damage when kept between 115% of titanium's yield strength (165 ksi) and 125% of titanium's yield strength (181 ksi). This should be used as a design consideration only for off-nominal short term operation of a bearing.	
Titanium races have extremely poor resistance to contact stress damage when kept above 125% of titanium's yield strength (181 ksi). This should not be used for design.	

Development of these guidelines was the primary goal of this effort. After sufficient data was gathered to establish these recommendations, leftover test samples were used for investigation into improving performance by enhancing bearing design. Preliminary results show coating the races using a pulse plasma nitriding process may significantly raise stress limits and reduce wear. This process exposes the subject material to plasma to introduce and diffuse nitrogen ions into the surface, improving the atomic structure. The coating also transforms the appearance of the titanium samples from a shiny silver color to a dull gold color. The result is an increased hardness at the surface without significant changes to part dimensions. Samples tested at contact stress values as high as 150% of titanium

yield stress did not show signs of race failure and generated less black dust than expected. While this is a promising result, additional testing must be done to verify performance and judge suitability for spacesuit applications.

## VII. Future Work

Future work is planned to evaluate the effects of 1) a more realistic cycle model, which includes changing directions during cycling and moving at slower cycling speeds, 2) additional materials combinations, such as the use of synergistic coatings, lubricants, and niton bearing balls, and 3) applications to more realistic space suit bearing geometries and the effects of non-uniform load distribution.

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